Renewable and Sustainable Energy Reviews 14 (2010) 478–485

Contents lists available at ScienceDirect

Renewable and Sustainable Energy Reviews

journal homepage: www.elsevier.com/locate/rser

Studies on pumping power in terms of pressure drop and heat transfer characteristics of compact plate-fin heat exchangers—A review

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ARTICLE INFO

Article history:
Received 3 January 2009
Accepted 11 June 2009

Keywords:
Compact heat exchangers
Offset fins
Wavy fins
Fluid flow non-uniformity
Turbulence

ABSTRACT

Renewable energy sources like solar energy, wind energy, etc. are profusely available without any limitation. Heat exchanger is a device to transfer the energy from one fluid to other fluid for many applications in buildings, industries and automotives. The optimum design of heat exchanger for minimum pumping power (i.e., minimum pressure drop) and efficient heat transfer is a great challenge in terms of energy savings point of view. This review focuses on the research and developments of compact offset and wavy plate-fin heat exchangers. The review is summarized under three major sections. They are offset fin characteristics, wavy fin characteristics and non-uniformity of the inlet fluid flow. The various research aspects relating to internal single phase flow studied in offset and wavy fins by the researchers are compared and summarized. Further, the works done on the non-uniformity of this fluid flow at the inlet of the compact heat exchangers are addressed and the methods available to minimize these effects are compared.

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1. Introduction

A plate-fin heat exchanger is a form of compact heat exchanger consisting of a block of alternating layers of corrugated fins and flat separators known as parting sheets. These heat exchangers can be made in a variety of materials such as aluminium, stainless

Abbreviations: FPI, fins per inch; OSF, offset fins; PHE, plate heat exchanger; PFHE, plate-fin heat exchanger.

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1364-0321/$ – see front matter © 2009 Published by Elsevier Ltd.
doi:10.1016/j.rser.2009.06.033

Nomenclature

A, a	wave amplitude (mm)
D_h	hydraulic diameter (m)
f	friction factor $(\Delta P \times g \times D_h)/(2 \times l \times V^2)$ dimensionless
g	acceleration due to gravity (m/s^2)
H, h	fin height (mm)
j	heat transfer coefficient ($St \times Pr^{2/3}$) dimensionless
L, L_f, l	plate/fin/offset length (mm)
l	total flow length of the fin (m)
l^*	Strip length dimensionless
Nu	Nusselt number
Pr	Prandtl number
R	wavy fin wavy tip radius (mm)
Re	Reynolds number based on hydraulic diameter
S, s	fin spacing (mm)
St	Stanton number
t	plate thickness (mm)
V	flow velocity (m/s)

Greek symbol

α	aspect ratio (S/H)
β	wall corrugation aspect ratio ($4a/\lambda$)
ΔP	differential pressure (Pa/m)
ε	channel spacing ratio ($S/2A$)
λ	wavelength (mm)

steels, nickel, copper, etc. depending upon the operating temperatures and pressures. They are widely used in aerospace, automobile and cryogenic industries due to its compactness (i.e., high heat transfer surface area-to-volume ratio) for desired thermal performance, resulting in reduced space, weight, support structure, footprint, energy requirement and cost. Depending on the application, various types of augmented heat transfer surfaces such as plain fins, wavy fins, offset strip fins, louvered fins and perforated fins are used. They have a high degree of surface compactness and substantial heat transfer enhancement obtained as a result of the periodic starting and development of laminar boundary layers over interrupted channels formed by the fins and their dissipation in the fin wakes. There is, of course, an associated increase in the pressure drop due to increased friction and form-drag contribution from the finite thickness of the interrupted fins. The surface geometries of wavy and OSF fins are described by the fin height (h), transverse spacing (s) and thickness (t). Interrupted flow length of the offset strip fin is described by offset strip/fin length (L) and for wavy fin, it is described by wavelength (λ). Thermo-hydraulic design of a compact heat exchanger is strongly dependent upon the performance of heat transfer surfaces (in terms of Colburn factor j and Fanning friction factor f vs. Reynolds number Re characteristics).

The major focus of the present review is the research work carried out in the field of offset strip fins and wavy fins. The studies on the effect of flow maldistribution, which arises especially in aerospace applications due to sharp changes in flow direction, are also highlighted. The review is summarized under the following broad sections.

- Offset fins characteristics.
- Wavy fins characteristics.
- Inlet fluid flow non-uniformity.

2. Heat transfer surface – offset and strip fins

The research works carried out on various aspects of offset fins as shown in Fig. 1 are summarized in this section.

2.1. Influence on flow and heat transfer

2.1.1. Fin thickness

The effect of larger fin thickness in offset fin heat exchangers has been reported in Kays [1], Patankar and Prakash [2] and Cur and Sparrow [3]. Kays [1] observed 25% increases in pressure drop for the fin thickness of 0.01 inch as compared to the 0.006 in. However, he has not reported the increase in heat transfer. According to Patankar and Prakash [2], for the Reynolds number range of 100–2000, f for the thick plate ($t/H = 0.3$) was 10–16 times the corresponding values for the zero-thickness plate and the heat transfer was increased with t/H, but not as much as one would expect from the increased average velocity based on the minimum free flow area and the increased surface area for the thick plates. The Stanton number values for the case of $t/H = 0.3$ were only about 24 times the corresponding values for the zero-thickness plate. But, according to Cur and Sparrow [3], at higher Reynolds number, the increase in fully developed Nusselt number was about 40% over the range from $t/L = 0.04$ to 0.12. Even larger increases were encountered at lower Reynolds number owing to the triggering transition from laminar to turbulent flow. They have not mentioned the percentage of increase in pressure drop with respect to thickness.

2.1.2. Two- and three-dimensional computations

Differences between 2D and 3D numerical results have been reported in Xi and Shah [4] and Suzuki et al. [5]. According to Xi and Shah [4], it was found that the effect of 3D geometry was similar in the laminar flow region ($Re_{3D} \leq 1600$), when compared to the 2D results. In the transition flow region, the use of the 2D computation was inaccurate to predict the flow and heat transfer performance for the OSF surfaces and hence the effect of the 3D geometry was significant. But, Suzuki et al. [5] compared the 2D results with in-house experimental results and found that the 2D computations were in quantitatively good agreement with the performance

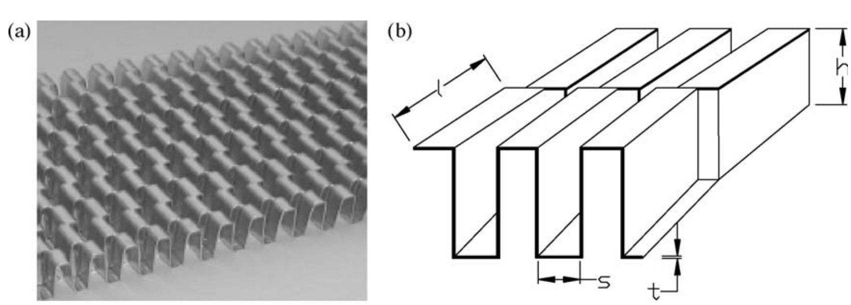


Fig. 1. Offset and strip fin: (a) photographic view and (b) dimensional notations.

study for the offset strip fin heat exchanger in the range of Re < 800. The range of Reynolds number quoted by both the authors is different. This may be due to the differences in geometrical dimensions considered in their analysis. Unlike the flow through the parallel rectangular duct, in the case of offset fin, the transition Reynolds number varies with geometrical dimensions of the fin.

2.1.3. Other working fluid

Studies on offset fins using liquid as working fluids have been reported in Sen Hu and Herrold [6,7] and Tinaut et al. [8]. Sen Hu and Herrold [6,7] compared liquids experimental data against air (Pr = 0.7) correlations from Joshi and Webb [9] and Wieting [10]. The results of both models and experiments were obtained for inlet fluid temperature of 10 °C. It was observed that at the same Reynolds number, the Colburn factor for liquids predicted using air models was approximately twice the Colburn factor obtained from the experimental data for the liquids. Air model over-predicted the heat transfer coefficient for liquids. This comparison between the air models and the experimental results demonstrated that the results for air cannot be accurately applied to liquid applications. Similarly, for a certain fin geometry and Reynolds number, the friction factor for different Prandtl numbers should be the same. But, the measured friction factor from liquid experiments was found to be higher than that from the air models. This difference might be due to burrs on the fins, created during the fin manufacturing process.

Based on their studies, it is very clear that the correlations developed for air medium cannot be directly used for liquid applications. So, there is need for the development of correlations, which can be directly applied for liquid medium applications. Hence, this review motivated the authors of present paper to carryout the research for developing the correlation for liquid applications.

2.1.4. Heat transfers and flow friction correlations

Development of heat transfer and pressure drop correlations in the form of j and f for the offset strip fins have been reported in London and Shah [11], Wieting [10], Webb and Joshi [9], Mochizuki and Yagi [12], Manson [13], Manglik and Bergles [14], Maiti [15], Muzychka et al. [16] and Ranganayakulu et al. [17]. These correlations are compared in Table 1.

Based on the above survey, it is understand that there are limited experimental data available in the open literature for different geometries of offset strip fins. Since, very few experimental data are available, the designers are forced to use available correlations or numerical tool for optimum design of compact heat exchangers. Now-a-days plenty of numerical tool are available in the market and different correlations are available in the open literatures. This reduces the experimental set up and testing cost to some extent. Even though plenty of correlations are available in the open literature for these type of fins, the designers are not able to select the best correlation, because of too much deviations observed among the correlations. These variations are shown in Fig. 2(a) and (b). It is evident that remarkable variations are present in the estimation of j and f factors of Offset strip surfaces. In the laminar region of f factors, all correlations are predicting well with experimental data of Kays and London experimental data [18] except Manson [13] correlation as given by Manglik and Bergles [14], which over-predicted by twice to thrice. In turbulent region of f factors, Maiti [15] and Manglik and Bergles [14] predicted well. But Webb and Joshi [9] and Mochizuki and Yagi [12] under predicted more than 15%. Weiting [10] over-predicted by 30% and Manson [13] over-predicted twice. In j factor is concerned for both regions, except Weiting [10] and Mochizuki and Yagi [12], all others under predicted by more than 20%. Manson [13] over-predicted by almost twice with experimental data of Kays and London [18]. Giving exact reasons for variation of these factors may not be possible due to involvement of so many parameters such as manufacturing aspects and testing conditions.

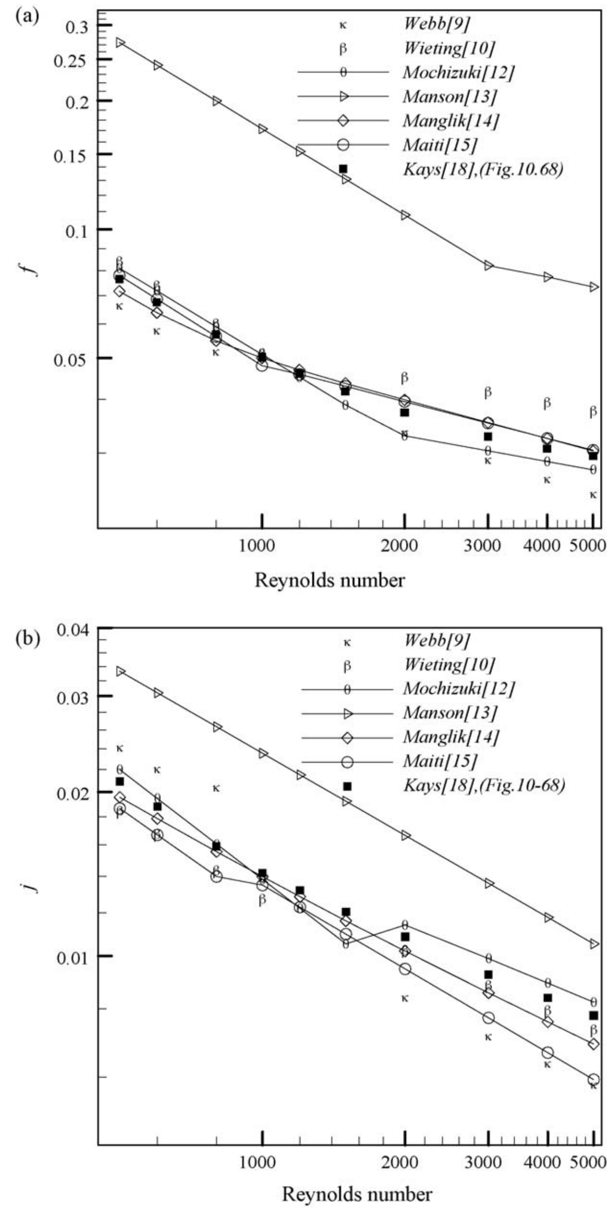


Fig. 2. Comparison of open literature correlations with Kays and London [18] experimental results for Fin : 1/8 - 16.00 (D) (a) f Vs Re and (b) j Vs Re.

3. Heat transfer surface – wavy fins

The various research and developmental activities involved in the area of wavy fins as shown in Fig. 3 are highlighted in this section.

3.1. Influence on flow and heat transfer

3.1.1. Comparison of wavy and parallel-plate channel

Comparison of heat transfer enhancement in the wavy channel with respect to a conventional parallel-plate channel has been reported in O'Brien et al. [19], Gradeck et al. [20], Wang and Vanka [21], Goldstein and Sparrow [22] and Sang Dong Hwang et al. [23]. O'Brien and Sparrow [19] reported that over the range of Reynolds

Table 1
Comparison of available j and f correlations for offset fins.

S no.	Authors	Correlations and hydraulic diameter
1	Wieting [10]	For $Re_D \leq 1000$ $f = 7.661(l/D)^{-0.384}\alpha^{-0.092}Re_D^{-0.712}$ $j = 0.483(l/D)^{-0.162}\alpha^{-0.184}Re_D^{-0.536}$ For $Re_D \geq 2000$ $f = 1.136(l/D)^{-0.781}(t/D)^{0.534}Re_D^{-0.198}$ $j = 0.242(l/D)^{-0.322}(t/D)^{0.089}Re_D^{-0.368}$ where $D = 2sh/(s+h)$
2	Webb and Joshi [9]	$Re \leq Re^*$ $j = 0.53(Re_D)^{-0.50}(l/D_h)^{-0.15}\alpha^{-0.14}$ $f = 8.12(Re_D)^{-0.74}(l/D_h)^{-0.41}\alpha^{-0.02}$ $Re \geq (Re^* + 1000)$ $j = 0.21(Re_D)^{-0.40}(l/D_h)^{-0.24}(t/D_h)^{0.02}$ $f = 1.12(Re_D)^{-0.36}(l/D_h)^{-0.65}(t/D_h)^{0.17}$ where $Re^* = 257(l/s)^{1.23}(t/l)^{0.58}D_h[t + 1.328(Re/lD_h)^{-0.5}]^{-1}$ $D_h = 2(s-t)h/[(s+h) + th/l]$
3	Mochizuki and Yagi [12]	$Re < 2000$ $j = 1.37(l/D_h)^{-0.25}\alpha^{-0.184}Re^{-0.67}$ $f = 5.55(l/D_h)^{-0.32}\alpha^{-0.092}Re^{-0.67}$ $Re \geq 2000$ $j = 1.17(l/D_h + 3.75)^{-1}(t/D_h)^{0.089}Re^{-0.36}$ $f = 0.83(l/D_h + 0.33)^{-0.5}(t/D_h)^{0.534}Re^{-0.20}$ where $D_h = 2sh/(s+h)$
4	Manson [13]	$j = \begin{cases} 0.6(1/D_h)^{0.5}Re^{0.6} & 1/D_h \leq 3.5 \\ 0.321\,Re^{0.5} & 1/D_h > 3.5 \end{cases}$ For $Re \leq 3500$ $f = \begin{cases} 11.8(1/D_h)\,Re^{0.67} & 1/D_h \leq 3.5 \\ 3.371\,Re^{0.67} & 1/D_h \leq 3.5 \end{cases}$ For $Re > 3500$ $f = \begin{cases} 0.38(1/D_h)\,Re^{0.24} & 1/D_h \leq 3.5 \\ 0.1086\,Re^{0.24} & 1/D_h > 3.5 \end{cases}$ where hydraulic diameter is defined by $D_h = 2sh/(s+h)$.
5	Manglik and Bergles [14]	$j = 0.6522Re^{-0.5403}\alpha^{-0.1541}\delta^{0.1499}\gamma^{-0.0678} \times [1 + 5.269 \times 10^{-5}Re^{1.340}\alpha^{0.504}\delta^{0.456}\gamma^{-1.055}]^{0.1}$ $f = 9.6243Re^{-0.7422}\alpha^{-0.1856}\delta^{0.3053}\gamma^{-0.2659} \times [1 + 7.669 \times 10^{-8}Re^{4.429}\alpha^{0.920}\delta^{3.767}\gamma^{-0.236}]^{0.1}$ where $D_h = (4shl/2(sl + hl + th) + ts)$, $\delta = t/s$ and $\gamma = t/l$
6	Maiti [15]	$Re < Re^*$ $f = 4..67(Re)^{-0.7}(h/s)^{0.196}(l/s)^{-0.181}(t/s)^{-0.104}$ $j = 0.36(Re)^{-0.51}(h/s)^{0.275}(l/s)^{-0.27}(t/s)^{-0.063}$ $Re > Re^*$ $f = 0.32(Re)^{-0.286}(h/s)^{0.221}(l/s)^{-0.185}(t/s)^{-0.023}$ $j = 0.18(Re)^{-0.42}(h/s)^{0.288}(l/s)^{-0.184}(t/s)^{-0.05}$ $Re^*_f = 648.3(h/s)^{-0.06}(l/s)^{0.1}(t/s)^{-0.196}$ $Re^*_f j = 1568.58(h/s)^{-0.217}(l/s)^{-1.433}(t/s)^{-0.217}$ Hydraulic diameter $= D_h = (2lh(s-t)/ls + hl + ht)$ $100 \leq Re \leq 10{,}000$
7	Muzychka and Yovanovich [16]	$f = [\{(fRe_{D_h}(d_h/D_h)/Re_{d_h}) + 1.328(Re_{d_h}(L_f/d_h))^{-1/2}\}^n + \{0.074(Re_{d_h}(L_f/d_h))^{-1/5} + ((Ht + (st/2))/2L_f(H+s))C_D\}^n]^{1/n}$ where C_D = form-drag = 0.88 and $1.3 < n < 5$. $j = [\{((Nu_{D_h}(d_h/D_h)/Re_{d_h}Pr^{1/3}))^5 + ((0.641(fRe_{D_h})/Re_{d_h}^{2/3})^{1/3}(d_h^2/D_hL_f)^{1/3})^5\}^m + \{0.037(Re_{d_h}(L_f/d_h))^{-1/5}\}^m]^{1/m}$ where D_h = Hydraulic diameter of the sub channel = $4A/P$. d_h = hydraulic diameter of OSF array = $4V_{free}/A_{wet}$. A = Area (m^2), V = volume (m^3) $2 < m < 5$, correlation is valid for all regimes.

number (1500–25,000), the enhancement of heat transfer as compared to a conventional parallel-plate channel was about a factor of 2.5. Friction factors obtained from axial pressure distributions were virtually independent of the Reynolds number and equal to 0.57, a value appreciably greater than that for unidirectional duct flows. Analog conclusions are reported in the works of Gradeck et al. [20]. They showed that the heat transfer enhancement of about 2 times for the range of Reynolds number (0–7500). However, Wang and Vanka [21] and Goldstein and Sparrow [22] suggested that in the steady-flow regime, the average Nusselt numbers for the wavy wall channel were only slightly larger than those for a parallel-plate channel. On the other hand, in the transitional and turbulent flow regimes, the enhancement of heat transfer was about a factor of 2.5. Friction factors for the wavy channel were about twice those for the parallel-plate channel in the steady-flow region, and remained almost constant in the transi-

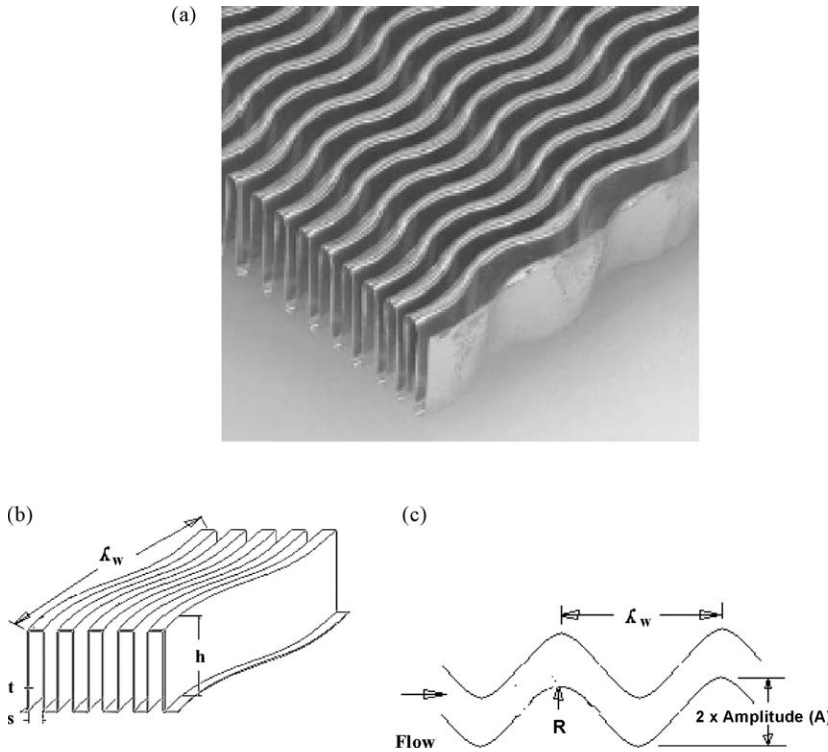


Fig. 3. Wavy fin (a) photographic view (b) and (c) dimensional notations.

tional regime. These authors statement is giving contradiction with O'Brien et al. [19], Gradeck et al. [20] at very low Reynolds number. According to Sang Dong Hwang et al. [23], the results showed that for the wavy duct, high performance factors of 2.2 were obtained at a low Reynolds number of 1000 due to relatively higher mass transfer enhancement than increase in pressure loss. But the performance factor was decreased gradually at $Re \geq 1000$, because the secondary vortices disappear and flow separation/reattachment flow characteristics occurred. These authors statement is more or less matching with O'Brien and Sparrow [19] and Gradeck et al. [20] at the low Reynolds number, but giving contradiction at high Reynolds numbers with all above authors. Hence, the performance study of wavy fin needs further research to eradicate these contradictions.

3.1.2. Turbulence models

Accuracy of the various turbulence models for the numerical simulation of wavy fin have been discussed in the works of Patel et al. [24], Amano et al. [25–27] and Manabu Horiuchi et al. [28]. According to Patel et al. [24], the two-layer turbulence model of Chen and Patel [29] appears to capture most of the important physical features of such flows. Another study (Amano [25]) revealed that the slope of the Nusselt number on the Reynolds number obtained using standard k–ε model with a special three-layer near-wall agrees with the experimental data of Izumi et al. [30]. However, the computed Nusselt numbers in the turbulent flow regime were 10–20% lower than the measured values. Their further studies (Amano et al. [26,27]) showed that the average Nusselt numbers computed by using the Reynolds-stress model (RSM) were about 20% higher than those obtained by using the k–ε model, and consequently agreed well with experimental data. This is because the turbulence level predicted in the recirculating region by the k–ε model was much lower than the results of Reynolds-stress model, thus resulting in lower heat transfer rates in the channel. In addition, the RSM can take nonisotropic effects into account, which were strong in the recirculation region. Later, Manabu Horiuchi et al. [28] made a numerical simulation of turbulent flow with the RANS turbulent models. The models were the two-layer, the low Reynolds number k–ε, and the V2F models. The simulation results were compared with those of the DNS and the prediction accuracy was discussed. The above comparisons reveal that sophisticated models such as the V2F model and the non-linear low k–ε model predicted more precisely the influence of flow in the recirculation region on the wavy wall.

3.1.3. Geometrical parameters

The effects of different non-dimensional geometrical parameters such as $\alpha = S/H$, $\beta = 4a/\lambda$, $\varepsilon = S/2A$ and $2A/H$ on heat transfer and pressure drop have been reported in Metwally and Manglik [31], Wang and Chen [32], Jiehai Zhang et al. [33], Manglik et al. [34] and Reza Motamed Ektesabi and Mitsuo [35]. The ranges of studies for these parameters are summarized in Table 2. The above survey reveals the following conclusions.

- The optimum (j/f) performance is obtained for corrugation geometries in the range $0.3 \leq \beta(4a/\lambda) \leq 0.6$. In the non-swirl flow regime, the enlarged surface area of the corrugated-plate is solely responsible for the enhancement and the thermal benefits are significantly less.
- The amplitude of the Nusselt number and skin friction factor are increased with an increase on the amplitude wavelength ratio. However, at a sufficiently larger value of amplitude wavelength ratio, the corrugated channel acts as an effective heat transfer device, especially, at higher Reynolds number.
- The heat transfer performance in the swirl flow regime ($Re \approx 600$) is found to be enhanced considerably for all β and $\varepsilon > 0.25$, when compared with the (j/f) performance of flat fins, and a peak performance is obtained with $1.0 < \varepsilon < 1.2$. In low flow rates ($Re \approx 10$), on the other hand, a much larger fin waviness severity ($\beta > 0.5$) may be required in order to achieve any significant enhancement.

Table 2
Range of parametric studies carried out on geometrical parameters of wavy fins.

Sl. no.	Authors	Range of study	Reynolds/Prandtl number
1	Metwally and Manglik [31]	$0<\gamma<1.0$ for $\varepsilon=1.0$.	$10<Re<1000$, $Pr=5$, 35, and 150
2	Wang and Chen [32]	$0<a/\lambda<0.5$.	$100<Re<700$, $Pr=0.71$ and 6.93
3	Zhang et al. [33]	$0.125\le\gamma\le0.5$ and $0.1\le\varepsilon\le3.0$.	$10<Re<1000$, $Pr=0.7$
4	Manglik et al. [34]	$0.240\le\alpha\le0.968$, $0.303\le\varepsilon\le1.22$ for constant $\gamma=0.2667$.	$10<Re<1000$, $Pr=0.7$
5	Reza Motamed Ektesabi and Mitsuo [35]	Fin height (H) = 3, 5, 10, 15 and 20 mm and wave amplitude ($2A$) = 10, 15, and 20 mm.	$300<Re<40{,}000$, $Pr=6.93$
6	Gschwind et al. [36]	1.8 mm $\le s\le$ 6.4 mm and $s/a=1$, 1.5, 2.5, 2.9, 3.2 and 3.5 for constant $\lambda=26$ mm, amplitude $a=1.825$ mm, the dimensionless wavelength $\lambda/a=14.25$, the duct length and width was 400 mm and 150 mm.	$50\le Re\le10{,}000$, with air and in a smaller range with water as working fluid. The Reynolds number (related to the duct heights)
7	Patel et al. [37]	$2a/\lambda=0.02$, 0.10, and 0.40. Where $2a$ and λ indicated the height and length of the cavity respectively.	$Re=8610$ and $Re=12{,}800$
8	Sparrow and Comb [38]	–	$2000<Re<27{,}000$, $4<Pr<12$.
9	Hang Scok Choi and Suzuki [39]	Wave amplitude was changed in three steps 0.01, 0.05 and 0.1.	$Re=6760$, $Pr=0.7$
10	Xie et al. [40]	Wave length = 4, 4.5 and 6 mm, wavy pitches = 9, 13 and 15 mm and channel width = 1.5, 2 and 3 mm.	$100<Re<1100$, $Pr=0.7$

- The cross-section aspect ratio (α) and fin separation (ε) appear to have competing effects on the thermal-hydraulic performance, as measured by the surface area goodness factor (j/f) or core compactness, and the optimum dependent upon the flow regime. Nevertheless, increasing fin density (or decreasing ε) tends to promote a relatively better (j/f) performance under swirl flow conditions and thus provide for a more compact wavy plate-fin heat exchanger core.
- At $2A/H<1$, the friction factor is increased somewhat due to the transition from laminar to turbulent flow. While at $2A/H>1$, it is decreased monotonically with increasing Reynolds number throughout the flow regime. Further, the formation of a large-scale vortex is the cause of the transition from laminar to turbulent flow.

The above authors studied about the net effects of dimensionless geometrical parameters on heat transfer and pressure drop. The individual effects of geometrical parameters such as fin height (H), fin amplitude (A), fin wavelength (λ) and fin spacing have been reported in Gschwind et al. [36], Patel et al. [37], Sparrow and Comb [38], Hang Scok Choi and Kenjiro Suzuki [39] and Gong-Nan et al. [40]. Based on their studies, they following conclusions are derived.

- It is possible to introduce longitudinal vortices in the wavy duct flows by centrifugal flow instability. The flow instability exists in a small range of Goertler numbers and depends strongly on the duct heights. Unstable duct flow over the whole duct length is observed only at small duct heights. The above mentioned results proved that the pressure loses in the channels with longitudinal vortices are small in comparison with those of turbulent duct flows.
- With Sparrow and Comb [38] duct configuration, the increase in the inter wall spacing gives rise to a 30% increase in the fully developed Nusselt number relative to that of Reza Motamed et al. [35], however the friction factor is increased more than twice.
- Finally, it is concluded that intensity of heat transfer may be improved with the increase of wave amplitude, or the decrease of wavelength, fin spacing and fin height.

3.1.4. Flow and heat transfer correlations

Development of correlations for wavy fins in the form of Nusselt number and fanning friction factor has been studied in Sparrow and Hossfeld [41], Stasiek [42] and Yar-Tsai Lin et al. [43]. The first two authors gave correlations under single phase flow conditions, whereas the last author gave correlation under two phase flow (humid) conditions. Their correlations are compared in Table 3. The Nusselt number correlations of first two authors have been compared with the experimental results of Gradeck et al. [20] and found too much variation among the results. This may be due to the difference in geometrical parameters considered in their analysis. Based on the review, it is found that very few correlations are available for wavy fins, when compared to offset fins. However, on correlations have been reported considering the effect of variations in geometrical parameters as considered in offset fin case. So, there is a need to develop the correlations by considering these parameters. At present, the authors of this paper are involved in the development of correlations using CFD tool FLUENT and the research is under progress.

Further, Sparrow and Hossfeld [41] has also studies the effects of the rounding of the corrugation on the heat transfer enhancement for a wide range of Reynolds number ($2000<Re<33{,}000$) and for $4<Pr<11$. They used water as working fluid. For the given flow parameters, they showed that the rounding induced a decrease of the Nusselt numbers due to the smoothing of flow. However, the effect of different values of wavy tip radius on flow and heat transfer is not highlighted anywhere. But, variations in wavy tip radius are having significant effects on flow and heat transfer. This motivated the present authors to carry out the study on the wavy tip radius effects and also inclusion of this parameter in the correlation development to increase the prediction accuracy of future correlations.

4. Flow non-uniformity analysis in the compact heat exchangers

The effects of flow non-uniformity and its reduction have been addressed by Ranganayakulu et al. [17,44], Wen et al. [45], Zhang and Yanzhong [46], Anjun et al. [47], Ranganayakulu et al. [48], Lalot et al. [49], Ranganayakulu et al. [50] and Ranganayakulu and

Table 3
Comparison of available correlations for wavy fins.

Sl no.	Authors	Correlations
1	Sparrow and Hosssfeld [41]	$Nu=0.491\times Re^{0.632}\times Pr^{0.3}$
2	Stasiek [42]	$Nu\propto Re^{2/3}$
3	Lin et al. [43] (For two phase flow i.e., under humid condition)	$Nu=0.02656\,Re_{Dh}^{0.92333}(F_s/D_h)^{2.5906}\theta^{0.47028}RH^{0.07773}$ $f=0.02403\,Re_{Dh}^{-0.41543}(F_s/D_h)^{-0.096529}\theta^{1.3385}RH^{-0.13035}$ Where F_s = fin spacing (m), θ = corrugation angle (°), RH = inlet relative humidity (%)

Table 4
Comparison of available methods to reduce the non-uniformity of flow.

Sl. no.	Authors	Method used to reduce flow non-uniformity	Type of heat Exchanger
1	Ranganayakulu et al. [17,44], 2007	Baffle plate and inlet pipe orientation	PFHE
2	Wen et al. [45], 2006	Baffle plate	PFHE
3	Zhang and Yanzhong [46], 2003	Two-stage distribution structure	PFHE
4	Anjun et al. [47], 2003	Second header installation	PFHE
5	Lalot et al. [49], 1999	Optimum location of baffle plate	PHE
6	Ranganayakulu et al. [50], 1997 and Ranganayakulu and Seetharamu [51], 1999	Improved design of existing header	PFHE

Seetharamu [51]. Ranganayakulu et al. [17] and Ranganayakulu et al. [44] analyzed two typical compact plate-fin heat exchangers using Computational Fluid Dynamics (CFD) simulation tool for quantification of flow maldistribution effects with ideal and real cases. In addition, suitable baffle plates designed to be placed at the inlet of the core for improvement in flow distribution based on the CFD study. Finally, they proved that flow non-uniformity has been drastically reduced due to the implementation of baffle plates. To the end, *j* and *f* data for some of the fins were presented.

Wen et al. [45] have investigated flow characteristics of flow field in the entrance of a plate-fin exchanger by means of Particle Image Velocimetry (PIV). Based on experiments, they suggested that punched baffle could effectively improve fluid flow distribution in the header.

Zhang and Yanzhong [46] have investigated the flow non-uniformity in a plate-fin heat exchanger by CFD software. Based on the investigation, two modified headers with a two-stage-distributing structure were proposed to reduce the flow non-uniformity. They proved that the fluid flow distribution in plate-fin heat exchangers (PFHE) is more uniform if the ratios of outlet and inlet equivalent diameters for both headers are equal.

Anjun et al. [47] introduced the concept of second header installation. By experimentation, they proved that the performance of flow distribution in PFHE was effectively improved by the optimum design of the both header configurations.

Ranganayakulu and Panigrahi [48] analyzed a cross flow two pass plate-fin compact heat exchanger, accounting for the effects of non-uniform inlet fluid flow due to headers using a finite element method. Using inlet flow non-uniformity models, the exchanger effectiveness and pressure drops and its deterioration due to the effects of flow non-uniformity were calculated for design case of heat exchanger. Based on the flow non-uniformity effects, they modified the exchanger hot side inlet and outlet headers to improve the exchanger effectiveness and reduce the pressure drop.

Lalot et al. [49] investigated the effect of flow non-uniformity on the performance of plate heat exchangers. They found the optimum location of perforated grid in the inlet header and observed reverse flow with poor header configuration. Most of previous research mainly investigated the effect of flow non-uniformity on the heat exchanger performance deterioration based on their specific flow maldistribution cases.

Ranganayakulu et al. [50] and Ranganayakulu and Seetharamu [51] investigated the effect of two-dimensional non-uniform flow distribution at inlet on both hot and cold fluid sides of cross flow plate-fin heat exchangers using a finite element model. It was found that the performance deteriorations and variations in pressure drops are quite significant in some typical applications due to fluid flow non-uniformity.

The methods adopted by the researchers to reduce the non-uniformity of fluid flow are compared in Table 4. However, the efficiency of the each method cannot be compared, because each researcher has used different geometry of the heat exchanger and the analysis was carried out at different Reynolds number. In order to choose the best method, the flow non-uniformity analysis is to be carried out on a typical heat exchanger at a particular Reynolds number by using all the available methods or combination of any two or three methods. By this way, it is definitely possible to improve the flow uniformity further at the inlet of the compact heat exchangers.

5. Conclusion

A review on the research and developments of offset and wavy compact plate-fin heat exchangers has been carried out. The information obtained is divided into three parts: offset fins, wavy fins and non-uniformity of inlet fluid flow. It is observed that researchers have generated plenty of experimental design data using air as working fluid. However, very few researchers have worked with other fluids. Further, based on the review, it is also found that even though plenty of correlations are available for offset fins, the designers may be confused to choose the best one due to the larger deviations between the correlations and experimental results. These variations are more than 30% for *f* factor and more than 20% in the case of *j* factor with respect to Kays and London experimental results [18].

In the case of wavy fin, the comparison of turbulence models studied by various researchers for the numerical simulation of wavy fin shows that sophisticated models such as the V2F model and the non-linear low k–ε model are predicting more precisely the influence of flow in the recirculation region on the wavy wall. It is observed that very few researchers have attempted at the development of correlations for wavy fins in the form of Nu and *f*. However no correlations have been reported considering the effect of variations in geometrical parameters and hence, the available correlations will lead to a lot of deviations with respect to experimental data. In addition to the above, it is also found that rounding of wavy fin corner decreases the Nusselt number due to the smoothing of flow. This indicates that wavy tip radius is having significant effects on flow and heat transfer. So, there is a need for further research in this aspect. Finally, the methods adapted by the various researchers to minimize the non-uniformity of flow are compared.

This review helps the researchers to carry out their further research in this field and also gives awareness for the designers to select the accurate design data (*f* and *j*) for the optimum design (i.e., minimum pumping power and efficient heat transfer) of compact heat exchangers. This optimum design in turn leads to energy savings in terms of cost.

Acknowledgment

The authors wish to acknowledge the Aeronautical Development Agency, Bangalore, for allowing to publish this paper.

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